

48 Thermoacoustics

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Abstract: Thermoacoustic heat engines offer mechanically simple energy conversion that can utilize a wide variety of heat sources – including solar energy, biomass, and even the “waste” heat from internal combustion engines and industrial processes. This chapter will address the gas thermodynamics that enable such machines and discuss the practical elements that comprise thermoacoustic machines that act either as a converter of heat energy to another form of energy (such as electrical or mechanical energy) or as a heat pump that “moves” heat from a cold region to a warmer one. The distinction between the two topologies of thermoacoustic machines, stack-based and regenerator-based, will also be clarified and the differences between the two made clear. Finally, the latter portion of the chapter will discuss existing and potential applications for thermoacoustic machines.

Introduction

At the end of the last century, the National Academy of Engineering created a list called the “Greatest Engineering Achievements of the Twentieth Century.” On that list, two of the top ten achievements have a heat engine at their heart: the automobile (rated second) and air-conditioning/refrigeration (rated tenth). After nearly 100 years of widespread adoption and significant incremental improvements to each of these products that have greatly improved the quality of life, the automobile engine and the cooling machine have also been identified as causing a large fraction of the atmospheric pollution problems. As a result, there is a significant global research and development effort underway to not only improve the efficiency of both automobile/truck engines and cooling machines, but also to design automobiles that do not use fossil fuels and refrigerators that do not require HFC/HCFC refrigerants that are 2,000–3,000 times more potent as a global warming gas than carbon dioxide. To address these challenges, novel heat engines are being developed that utilize thermoacoustic energy conversion.

Thermoacoustic energy conversion technology can be a part of a strategy to reduce global warming gas emissions for four primary reasons:

- The working fluid of these machines is typically an inert gas, like helium, which has no global warming potential (GWP).
- The efficiency is comparable with existing power conversion technology and has room for growth.
- The machines are fuel-flexible and do not require fossil-fuel based liquids.
- The machines are mechanically simple, which makes them an attractive target for commercialization.

Thermoacoustic machines belong to the closed-cycle class of heat engines and are subject to Carnot efficiency restrictions. They are similar in many ways with Stirling machines. The goal for the technology development has been to create a closed-cycle engine technology that is both efficient enough to present significant fuel savings (and therefore reduce emissions) while also being mechanically simple enough to meet price

and reliability targets. A schematic of a thermoacoustic refrigerator compared to a vapor-compression machine is shown in  Fig. 48.1.

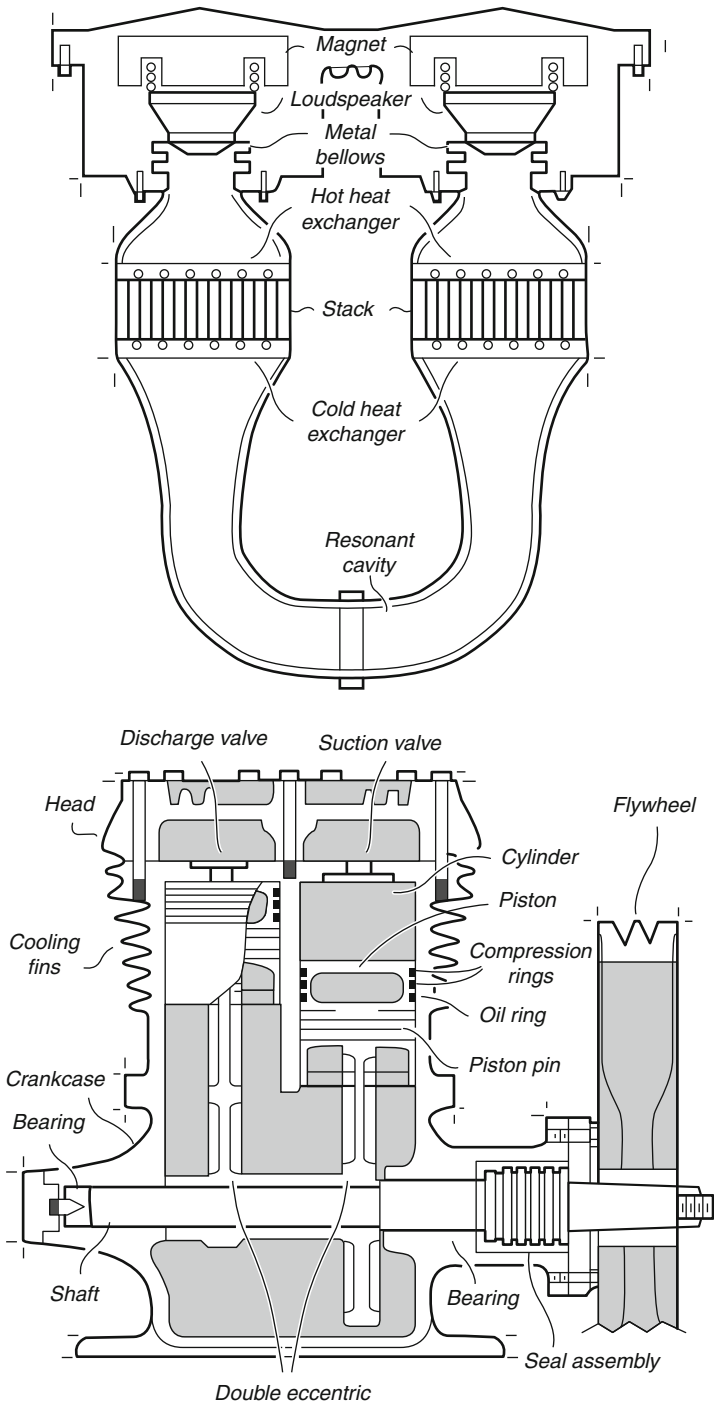
This technology has been under active development since the late 1970s with the work of Peter Ceperley [1] and, independently, John Wheatley, Greg Swift, and collaborators [2]. In the three decades that have elapsed since the foundations were laid, the physical understanding of the hydro- and thermodynamics in these devices have been well developed and codified largely by Greg Swift in a textbook [3] and, previously, in a comprehensive journal article [4]. This chapter will not attempt to recast or even improve upon this work; readers who have an interest in the detailed physics of thermoacoustic heat engines will find satisfaction in Swift's textbook treatment of the subject.

This chapter will provide an explanation of the two main types of thermoacoustic heat engines (stack-based and regenerator-based). Some of the applications to which thermoacoustic technology has seen development efforts will be discussed followed by an outline of some of the challenges that have been discovered that pose hurdles to adoption on the scale that could have an impact on the mitigation of climate change.

Thermoacoustic machines can be broken into two broad application fields: machines that convert the flow of heat from a hot temperature to a colder one into sinusoidal oscillations in a gas (acoustic waves), and machines that use an acoustic wave as an input to “pump” heat from a cold temperature to a hotter one. Machines designed to accomplish the former are typically called engines, and they require some small oscillation of the gas to start an instability that grows and is sustained by the temperature difference and continuing flow of heat from hot to cold. In practice, the small oscillation required to trigger the instability occurs naturally from convection currents of the gas. Machines designed to utilize the energy in a sound wave to pump heat from a cold temperature to a warmer one are called alternately refrigerators, heat pumps and sometimes air-conditioners when targeted toward indoor climate control. In the case of the thermoacoustic heat pump, a source of sound is required: often this is accomplished using a transducer (e.g., linear motor) but the sound could also be provided by a separate (but coupled) thermoacoustic engine.

At the core of the machine is a pair of heat exchangers which bookend a slab of porous media. The working gas of the machine can flow through the heat exchangers and through the porous media between them. As the gas in thermal contact with these core elements experiences pressure and velocity changes with the correct phase, the desired effect (either amplification of the oscillations or heat pumping) is realized.

This core is mounted into a container of carefully designed dimensions so that the gas oscillations will be resonant, which not only sets the frequency of the gas oscillation, but provides an impedance (the ratio of the oscillating gas pressure to the oscillating gas velocity) enhancement favorable to the design and efficiency of the machine. From the standpoint of efficiency, a high impedance is preferable because the gas velocities are small for a given power flow (power in this case is the product of the oscillating pressure amplitude, the oscillating velocity amplitude and the phase between them). The loss in the porous medium tends to be attributable more to drag than to non-adiabatic pressure oscillations since the pores have to be fairly small to get good thermal communication



■ Fig. 48.1
(Continued)

between the solid and the gas. Additionally, for heat pumps that are driven electromechanically, a high impedance means that a relatively small back-and-forth motion of the piston is enough to make large pressure swings: small piston motions can take advantage of many kinds of seals that do not require lubrication, like flexure seals or dry-fit clearance seals.

Thermoacoustic machines are also categorized on another criterion: the pore size of porous medium that is placed between the two primary heat exchangers mentioned above. One category uses a porous medium called a “stack,” the other category uses a medium called a “regenerator.” The fundamental difference between the two is the size of the pores. This difference will be discussed later the chapter. The regenerator-based machines have the potential to be more efficient than the stack-based machines at the expense of slightly more complication. Each type of machine may have a role in particular applications.

A final element shared by many thermoacoustic machines is a transducer of some kind that either converts electricity to sound or the reverse, converting sound waves produced by an engine to electricity. A linear motor is a common device that can perform both of these functions; there are linear motors with moving magnets [5] which are typically preferred for reliability due to the coils being stationary. Moving coil designs have also been used, especially for research purposes (these are similar to a loudspeaker in design).

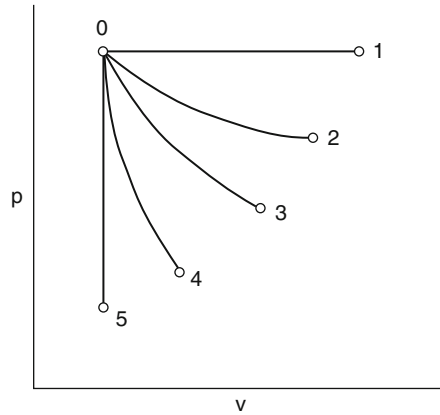
Heat Engine Cycles Background

A machine that either converts heat flow into mechanical work or forces heat to flow from a cold object to a hot one using mechanical work can be described in terms of the thermodynamic steps that make up one cycle of operation. The state of the working medium at the end of each step can be predicted (if the state before the change is known) using the expression $p_0 V_0^\gamma = p_n V_n^\gamma$, where γ is a variable that depends upon the details of the thermodynamic process by which the system changed state. Thermodynamic state change is commonly categorized in five ways listed below and shown on a p - v diagram in [Fig. 48.2](#):

1. Isobaric (constant pressure, $\gamma = 0$)
2. Isothermal (constant temperature, $\gamma = 1$)

■ Fig. 48.1

The schematic of a thermoacoustic refrigerator is shown on the *left* which cooled radar electronics aboard the USS Deyo in 1995. A pair of linear motors convert electrical energy into a sound wave which fills the resonant cavity where cold heat exchangers allow heat from the cold radar electronics to enter the cavity and be pumped to the hot heat exchangers which exhaust that heat to the ambient temperature environment. The schematic of the thermoacoustic machine is somewhat mechanically simpler than the schematic of the vapor compression machine shown on the *right* which requires lubricated sliding/rotating seals and often relies on working fluids that have a high global warming potential (Artwork by Tom Dunne courtesy of The American Scientist, used with permission)



■ Fig. 48.2

The four special cases of polytropic state change are shown on a pV diagram. For an ideal gas, γ is the ratio of specific heats for the adiabatic case. When considering an ideal gas, there are no physically meaningful values of γ between the ratio of specific heats and ∞ (the constant volume case). The numbers correspond to the list of thermodynamic state change categories above

3. Polytropic ($1 < \gamma < c_p/c_v$)
4. Adiabatic (constant entropy, $\gamma = c_p/c_v$)
5. Isometric (constant volume, $\gamma = \infty$)

A heat engine cycle generally has four steps executed in succession. These four special cases of a polytropic process (and the generic case labeled “polytropic”) can be combined to realize ten different heat engine cycles since only two unique processes can be included in a cycle. (One reason for the requirement that each process must occur twice per cycle is that a machine built to execute a heat engine cycle must repeat its motion; for example, if a machine executes a constant volume process during one-quarter of the cycle, it will (usually) mechanically execute another one to return to the initial state.) These five thermodynamic processes can be combined into ten identifiable elementary heat engine cycles, which are listed in ► [Table 48.1](#) [6].

Rev. Robert Stirling’s contribution in 1816 to the development of heat engines was to recognize the importance of the regenerator (he called it the “economizer”) to increase efficiency, which was important as the price of coal had begun to rise steadily since the beginning of the Industrial Revolution [7, 8].

How a Regenerator Is Utilized to Pump Heat Efficiently

Since regenerator-based thermoacoustic machines are the first choice for efficiency, it makes sense to lay out the phenomenological physics of how an acoustic wave passing within it pumps heat and how it relates to the Stirling cycle. Following ► [Fig. 48.2](#), the pV

■ Table 48.1

A chronological list of ten elementary heat engine cycles showing that the concept of a regenerator occurred early in the development of heat engines

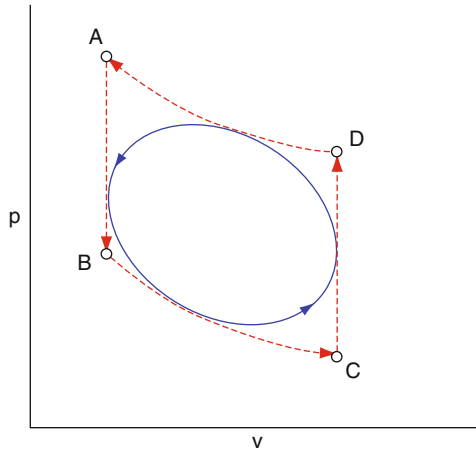
Type of state change	Credit	Year
1 & 5, isobaric and isometric	Papin	1690
1 & 3, isobaric and polytropic	Cayley	1807
2 & 5, isothermal and isometric	Stirling	1816
2 & 4, isothermal and adiabatic	Carnot	1824
2 & 3, isothermal and polytropic	Reitlinger	1843
1 & 4, isobaric and adiabatic	Joule	1852
2 & 1, isobaric and isothermal	Ericsson	1853
4 & 5, adiabatic and isometric	Otto	1867
3 & 4, polytropic and adiabatic	Lorenz	1894
3 & 5, polytropic and isometric	Crossley	1896

diagram for the Stirling cycle is shown in ► Fig. 48.3. A schematic of a machine that can execute this cycle is shown in ► Fig. 48.4.

The schematic diagram of the alpha-Stirling refrigerator shown in ► Fig. 48.4 contains a regenerator sandwiched between heat exchangers with two pistons, one on either side of the regenerator (these will be referred to as the cold- and hot-side piston in this discussion). At steady state, the ideal regenerator has a linear temperature gradient across its length. The pistons are connected by some linkage that provides for the correct relative motion of the pistons to make the machine function as a refrigerator or an engine (or nothing useful). One usual simplification is to suppose that each piston has the range of motion to be able to sweep out the entire volume of its cylinder, right up to the edge of the heat-exchanger/regenerator edge. The simplest view of the operation of this machine as a refrigerator contains four distinct strokes in the following order:

A–B Displacement: Both pistons move in concert to move the gas from the hot-side piston cylinder through the regenerator to the cold piston cylinder. The gas is cooled reversibly as it passes through the regenerator since the gas is assumed to be in perfect thermal contact with the regenerator material due to the porous design of the regenerator. Since the gas is colder, heat is (reversibly) absorbed by the regenerator from the gas. This movement through the regenerator is carried out so that the volume that the gas occupies is constant throughout the step. Since the temperature of the gas decreases at constant volume, the pressure of the gas must decrease during this step. Although no $p \cdot \Delta V$ work is done on or by the gas (since the volume did not change), the energy lost by the gas from thermal conduction can be expressed by the decrease in pressure at constant volume.

B–C Cooling: Now that the gas is at a low temperature, but at the starting volume, the cold-side piston moves away from the hot-side piston (the separation distance increases) which decreases the pressure of the gas even further. This expansion would cause the temperature of the gas to decrease to a temperature below the coldest temperature

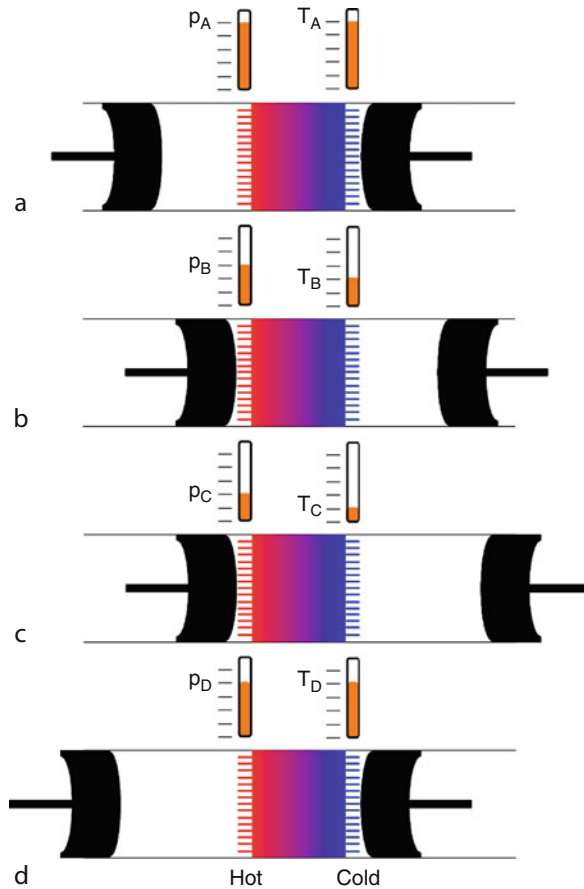


■ Fig. 48.3

A qualitative graph of pressure vs. volume for an articulated Stirling cycle. The oval (solid curve) represents the cycle driven with sinusoidal pressure and velocity oscillations. The outer curves (dotted) represent an articulated cycle

of the regenerator if the step were adiabatic. In the isothermal limit, during each differential step of this expansion stroke, the gas gets a differential amount colder which allows a differential amount of heat to move from the heat exchanger located at the cold end of the regenerator to the gas. The cold (or load) heat exchanger is assumed to have the same temperature as the cold end of the regenerator and, like the regenerator, this heat exchanger is in perfect thermal contact with the gas. Although the ideal Stirling refrigerator exhibits an isothermal expansion during this step, in real machines, this step is closer to an adiabatic expansion, as noted below. The job of the load heat exchanger is to move the heat from the thing to be cooled (e.g., ice-cream, medicine, foodstuffs) into the cold-side cylinder so that it can be transferred to the gas during this cooling step.

C–D Displacement: Following the cooling stroke the pistons move together to drive the gas back through the regenerator toward the hot-side piston. During this constant volume displacement, the gas is reversibly warmed to the hot-side regenerator temperature. Since the warming occurs at a constant volume, the pressure of the gas increases during this step. It is during this step that the parcel of gas recovers the energy given to the regenerator from step $[A] - [C]$. It is in this reversible recovery of energy that the regenerator proves its worth; because of the regenerator, the expansion stroke only has to move heat *at the cold temperature* and does not have to cool the charge of gas from ambient temperature to the cold temperature since the displacement stroke through the regenerator took care of that part. It is this idea of precooling and preheating the gas in preparation for heat transfer to or from a load that makes the Stirling cycle and other regenerative heat engine cycles useful. The fact that the regenerator can, in principle, do this precooling and preheating reversibly makes it efficient.



■ Fig. 48.4

This figure shows the piston motion in an alpha-Stirling refrigerator (the alpha configuration is where two distinct pistons are located in separate cylinders). The letter designations correspond to the pV diagram in ● Fig. 48.3, and the meters above each picture display the relative magnitudes of the gas pressure and average gas temperature

D–A Exhaust: Once all of the gas has been reversibly warmed to the hot-side temperature, the hot-side piston moves toward the cold-side piston compressing the gas that has been warmed. If isothermal, the amount of heat that was removed from the load exchanger plus the work done to remove it is absorbed by the exhaust heat exchanger. The job of the exhaust exchanger is to provide a path for that heat to leave the hot-side cylinder.

After reading this simplified step-by-step description of the operation some complications might be evident. First, a machine cannot produce such articulated motions to be able to easily mark, for example, the end of the cooling step and the beginning of the displacement step. In real machines, the pistons usually move in a sinusoidal motion

(as shown in ► Fig. 48.3), which not only blurs the distinction between each of the four steps, but also reduces the power density of the machine for a given operating pressure range. This reduction in power density can easily be understood by appreciating that the area enclosed by the lines of the pV diagram is the amount of work required to move heat from material at the cold temperature to the environment. If a machine was operating at a constant efficiency, it would be able to move more heat for a specified pressure and volume oscillation amplitude if it could operate on the dotted curves in ► Fig. 48.3 (articulated piston motion) than it could if the pistons moved with sinusoidal motion as shown on the ellipse.

Secondly, it would be difficult to make a machine that could execute isothermal expansions and contractions in the cylinder spaces on the left and right of the regenerator in ► Fig. 48.4. If the relative scale of the schematic shown in ► Fig. 48.4 were used to build a machine, that machine would be better modeled with adiabatic compressions and expansions in place of the isothermal ones described above. These adiabatic compressions decrease performance below the Carnot limit since during the adiabatic pressure change, the gas gets measurably warmer or colder than the regenerator end-temperatures. Consequently, when this hotter or colder gas enters the regenerator (or heat exchangers), heat moves across a finite temperature difference, which is an irreversibility that decreases maximum theoretical performance below the Second Law limit. Also called the Carnot limit, the Second Law of Thermodynamics sets an upper limit for the amount of heat that can be moved from a mass at a cold temperature and exhausted to the environment at ambient temperature for a given amount of input work. This limit depends only upon the cold temperature, T_c , and the ambient temperature, T_0 , and is expressed as $COP_C = Q_C/W = \frac{T_c}{T_0 - T_c}$, where Q_C is the amount of heat removed from the cold object and W is the amount of work required to move that heat (COP is an acronym that stands for coefficient-of-performance).

The Stirling cycle, as presented with isothermal compressions and expansions is reversible (in the limit of zero viscosity) and, therefore, has the theoretical potential to reach the Carnot limit of efficiency. In earlier days of Stirling machine design, assuming isothermal expansions and compressions made the analysis of potential designs tractable in closed form; the first published isothermal analysis of the machine was in 1871 by Gustav Schmidt, a professor at the German Polytechnical Institute at Prague. It was Schmidt, 50 years after the invention of the Stirling machine, who first represented the cycle with isothermal expansions and contractions because that model yielded a closed-form solution. Almost 100 years later, Finkelstein published the first analysis of a Stirling machine with non-isothermal expansions and compressions [7].

The Function of the Regenerator and Its Advantages Compared to a Stack

The purpose of the regenerator in Stirling machines is to provide for the temporary storage of heat for the oscillating working gas. The gas must be brought to the cold

temperature before the cooling (expansion) stroke is executed and then warmed before the exhaust (compression) is executed. The amount of heat removed to provide the cool gas for the expansion stroke is exactly the same amount of heat that must be returned to the gas in preparation for the exhaust stroke. The regenerator provides a place for the gas to temporarily store this heat; to do so, it must have enough heat capacity to be able to store the required heat to get the gas to the desired temperature and it also must be as thermally nonconducting as possible to reduce heat flow along its length, a heat flow that reduces the effectiveness of the regenerator. For this reason, many regenerators are made of stacked stainless steel screens which provide plenty of heat capacity and a large array of pore size possibilities. Since each layer of screen has a significant amount of thermal contact resistance, heat conduction along the length is minimized to an acceptable level.

Stack-Based Thermoacoustics

In the last 20 years of the twentieth century, much research and design was dedicated to thermoacoustic machines that utilized an oscillating gas with nominally standing wave phasing (gas pressure and velocity 90° out of phase) in the presence of a “stack” of closely spaced, rather nonconducting plates [4]. This stack of plates was designed with a plate-to-plate spacing on the order of a few thermal penetration depths, or the acoustic thermal boundary layer thickness. This acoustic boundary layer is called a “penetration depth” because it is the distance that heat (or momentum, in which case, it is referred to as the viscous penetration depth) can diffuse in one acoustic cycle. The penetration depths are defined as

$$\delta_\kappa = \sqrt{\frac{2k}{\rho_m c_p \omega}}$$

$$\delta_\nu = \sqrt{\frac{2\mu}{\rho_m \omega}}$$

where ω is the angular frequency of the gas oscillation, ρ_m is the mean gas density, c_p is the gas specific heat at constant pressure, k is the thermal conductivity, and μ is the shear viscosity of the gas. Since the stack plate-to-plate spacing might be twice this thermal penetration depth, the rate of heat transfer between the plates and a gas parcel translating in a plane equidistant from two plates is not fast compared to an acoustic period, as it might be for a parcel of gas near to the plates. The lag of heat transfer with respect to the motion of the active gas parcels (those parcels that are not too near the plates) is what allows the gas to execute a useful heat engine cycle – one that either spontaneously converts heat input into sound energy or one that uses sound energy to move heat from a cold reservoir to an ambient temperature one.

This natural phasing idea, utilized by designing the stack with the proper plate spacing for a particular thermal penetration depth (which depends on working gas properties and frequency of oscillation) is considered an innovation since it rids a machine of any number of complicated methods to impose the required phase difference between heat

transfer and gas motion. For example, in an automotive engine, each cylinder has at least two valves that must be opened and closed at the right time via the motion of rockers, cams, and pushrods. Most Stirling machines, as well as other air and gas engines, also have complicated linkages that impose the phasing of gas motion, pressure and heat input, and exhaust. However, the mechanical simplicity that comes with the exploitation of natural phasing brings with it a significant penalty in efficiency of the cycle. The fact that the thermal contact between the stack and the gas in the stack is not perfect (in order to get the right “natural” phasing) the heat that is exchanged between the stack and the gas is not exchanged in a reversible way [9]. Since this heat transfer that occurs each cycle happens over a finite temperature difference, the *theoretical* limit in efficiency of the stack-based thermodynamic cycle is lower than the Second Law limit.

Regenerator-Based Thermoacoustics

As opposed to using a stack as the second thermodynamic medium, the use of a regenerator allows the theoretical thermodynamic efficiency to reach the Carnot limit. The function of a regenerator is slightly different than that of a stack. While both the stack and regenerator provide heat capacity for the temporary storage of heat, the details of how this heat is exchanged with the gas are different. In the stack, this exchange is irreversible because the active gas is one or a few thermal penetration depths away from the stack material (as explained above). It is this irreversibility that is the source of the natural phasing: a simplicity that can be desirable and worth the efficiency penalty. In a regenerator, the gas passage pores are much tighter. This ensures that all of the gas in the regenerator is well within a fraction of a thermal penetration depth so that the thermal diffusion time is significantly shorter than an acoustic period. (A rule of thumb might be that the pore sizes are one fifth of the size of a thermal penetration depth.) Consequently, the transfer of heat between the regenerator and the gas is theoretically reversible which allows the efficiency to approach the Carnot limit. (However, since the pore size is small, the COP of any real incarnation of a regenerator-based heat engine will be limited by viscous loss in the regenerator, as well as other irreversibilities. In the comparison of stack-based to regenerator-based machines, it is somewhat arbitrary to compare the limiting case of a working fluid with zero viscosity since its effect is much bigger in the regenerator than the stack.) This gain in potential efficiency comes with the loss of the elegant natural phasing found within the pores of the stack. Therefore, the phasing of heat transfer and gas motion in the regenerator must be imposed by some other mechanism.

Using a Regenerator in an Acoustic Heat Engine

Since the natural phasing found in the stack is lost when taking advantage of the improved properties of the regenerator-based Stirling cycle, the correct phasing of heat transfer and gas motion must be imposed in some other way. In the late 1970s, Ceperley [1] recognized

that a traveling acoustic wave exhibits the correct phasing to force the oscillating gas to execute a Stirling cycle. This phasing, where oscillating pressure is in phase with the velocity of the gas is enforced in conventional kinematic Stirling machines by way of the mechanical linkage. Ceperley built an acoustic-Stirling engine apparatus that included a regenerator inside of a pipe that was several wavelengths long, but his engines failed to produce acoustic energy. Although the regenerator was providing the needed energy storage, it also presented a significant viscous loss to the acoustic wave passing through the regenerator matrix. This attenuation, proportional to the square of the acoustic velocity, was much too large to allow any acoustic pressure oscillations to build in the tube; as the pressure would increase from the Stirling cycle taking place in the regenerator, the acoustic particle velocity would also increase, causing increased attenuation in the regenerator which robbed the wave of the power produced by phased heat transfer in the regenerator.

This is best understood by considering that the impedance (the complex quotient of oscillating pressure divided by the gas velocity) of the wave traveling along the tube that is several wavelengths long is simply the characteristic impedance of the gas in the tube: the product of density and sound speed. This impedance is relatively low compared to the impedance found near the closed end of a tube that is half of a wavelength long. In such a tube, the magnitude of the impedance near the closed end is large compared to the characteristic impedance of that gas; the oscillating pressure is large in magnitude compared to the relative magnitude of the gas velocity which becomes zero where the gas meets the capped end of the tube. This location would be a good place to locate a regenerator, except for the fact that in such a standing wave tube, the pressure is almost 90° out of phase with the gas velocity.

This preference for high impedance (ratio of driving potential to flow velocity) is also found in the electrical power transmission network. Electrical power (alternating current) is the real part of the product of the complex voltage (analogous to pressure) and complex current (analogous to acoustic velocity) in the form $\dot{E}_{elec} = \frac{1}{2} \Re[\tilde{e} \cdot \tilde{i}^*]$, where the asterisk represents the complex conjugate. The energy lost to Joule heating, which is the dominant loss mechanism in power transmission, is the product of the square of the current and the electrical resistance, $\dot{E}_{loss} = i^2 R$, of the transmission line. Therefore, electrical power is transmitted at much higher voltage amplitudes than people like to use in their homes and at a significantly reduced current amplitude to reduce the resistive loss. Since power is the product of the potential and the flow, the amount of power transmitted can be maintained while the potential is increased and flow decreased proportionally to minimize losses.

In a regenerator-based acoustic machine, it is desirable to deliver a particular amount of power to the regenerator with the acoustic wave. The delivery of this power must have pressure and velocity in phase and at an acceptably large impedance so that viscous losses in the regenerator do not make the machine so inefficient as to be of no practical use. The first machines to successfully exploit this idea of an acoustic Stirling machine with the regenerator at a high impedance location was an engine built at Los Alamos by Scott Backhaus and Greg Swift [10] and an engine of Kees de Blok and collaborators in the Netherlands [11]. Their pioneering concept for

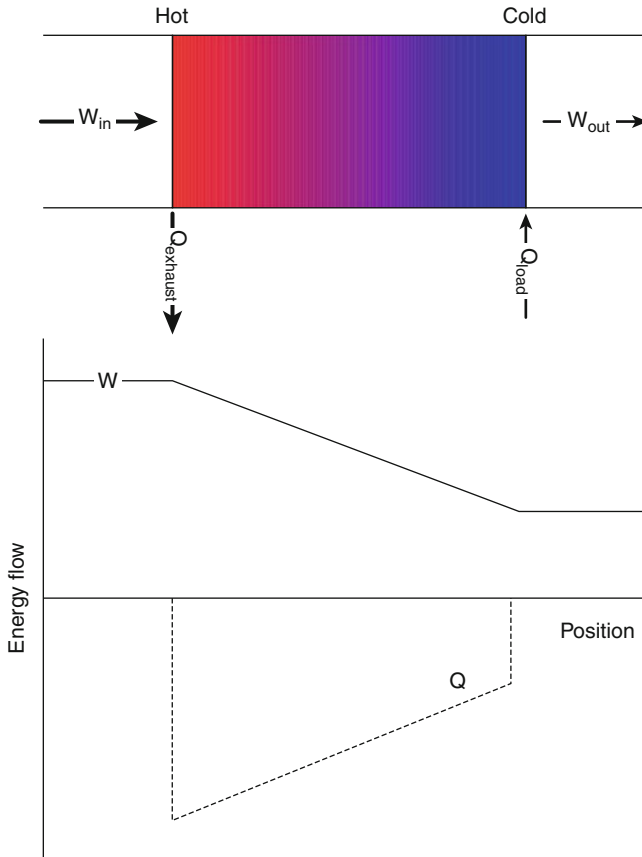
a thermoacoustic Stirling engine led the way for a new decade of thermoacoustic machine development that continues to this day.

The Acoustic Phasing Network

For successful utilization of a regenerator, several requirements must be met. First, pressure oscillations and velocity oscillations must be substantially in phase. The frequency of this oscillation must be low enough that the acoustic boundary layer thickness is several times larger than the average pore size of the regenerator matrix. The magnitude of the complex ratio of oscillating pressure to oscillating velocity must be relatively large in comparison to the characteristic impedance of the working gas. In addition to these elements, the energy flow in the machine is important.

In a Stirling refrigerator, the gas in the regenerator is a sink for mechanical (or acoustical) energy that passes from the exhaust (ambient temperature) side to the load (cold temperature) side. Energy in the form of mechanical work is supplied to the gas in the regenerator (see ● Fig. 48.5), and that gas absorbs some amount of this work in order to move heat against a temperature gradient from the cold end to the ambient (or hot) end of the regenerator. In a regenerator that transfers heat reversibly to a gas with no viscosity (this is a “perfect” regenerator and could also be thought of as one that generates no entropy), the remaining amount of work that flows out of the cold end of the regenerator is equal to the amount of heat removed from the load and pumped to ambient temperature. In this perfect regenerator, the amount of total energy at any location along the regenerator’s axis is zero. Since total energy or enthalpy, $\dot{H}_2$, is the sum of the work energy and the heat energy in the regenerator (and would include other types of energy if the magnitude was changing within the regenerator), the heat moving to the left (“up” the temperature gradient) is equivalent to the magnitude of work energy that is flowing to the right (the direction of propagation of the acoustic wave). In a machine with a perfect regenerator, in order to move an amount of heat, δQ per cycle some amount of work δW is absorbed per cycle in the regenerator as demanded by Carnot.

The work required by the Second Law of Thermodynamics to move an amount of heat δQ is related only to the temperatures between which the heat is moved T_0 and T_C : $\delta W = \delta Q \frac{T_0 - T_C}{T_C}$. In a real regenerator, the amount of work absorbed in the regenerator due to viscosity is greater than the Carnot minimum; since this work does not contribute to moving any useful heat, this extra work raises the total power in the regenerator to a value greater than zero. Likewise, since a real regenerator allows some conduction of heat from the ambient side to the cold (load) side and heat exchange in the real regenerator has irreversibility associated with it, the net amount of heat moved from the load to the ambient reservoir will not be as great as it could have been for the amount of work flowing into the regenerator. This reduction in the useful amount of heat moved “up” the thermal gradient within the regenerator also increases the magnitude of total power in the regenerator above the ideal value of zero. The concept of total power allows a quantitative way to judge the “effectiveness” of the regenerator and allows the designer



■ Fig. 48.5

This graph shows the energy flow through a regenerator that produces refrigeration. The *dotted line* represents the heat and the *solid line* shows the mechanical work. Note that everywhere in the regenerator, the sum of the heat and work is zero. The useful cooling capacity in an ideal regenerator is equal in magnitude to the amount of work that flows out of the regenerator at the cold (load) side

to know if a proposed change to the design of the machine or regenerator parameters will increase efficiency.

Within the Stirling community, machines are commonly classified into two major categories: kinematic and free-piston. In a kinematic machine, the two moving elements (either a piston and a displacer or two pistons) are connected together by a mechanical linkage and the motion of the two elements with respect to one another is completely defined by the mechanics of the linkage. Free-piston machines are a much more recent development, largely credited to Beale [12], but could also be attributed to Ringbom [8]. The recognition that the mass of the displacer could be made to resonate against the stiffness of trapped gas in the machine created a renewed interest in Stirling machines in

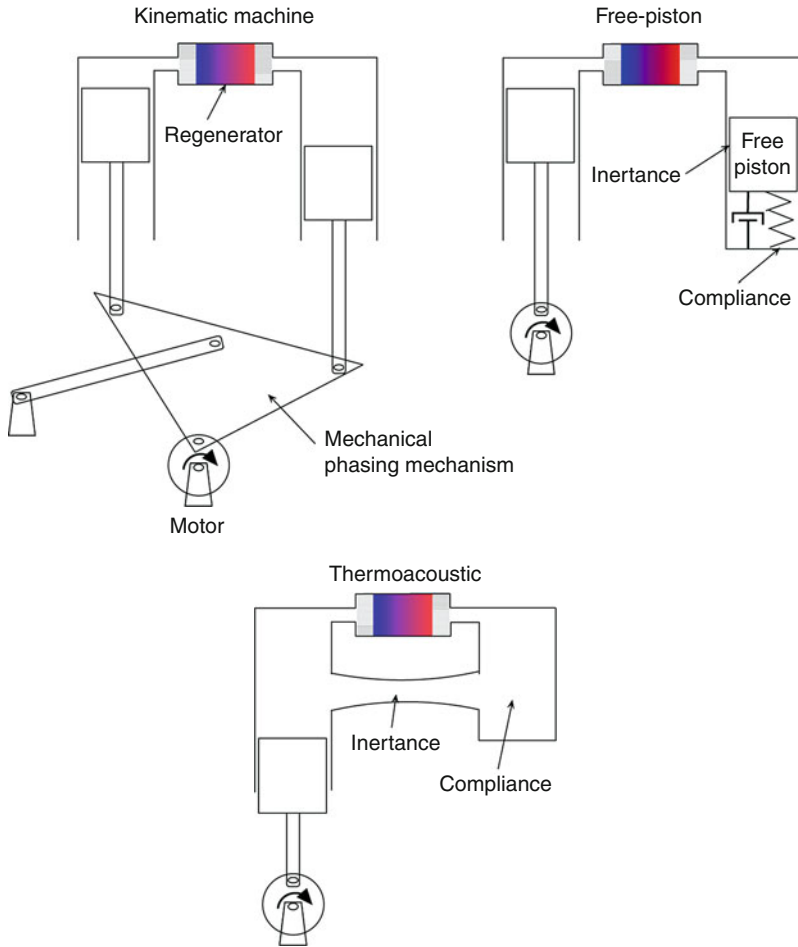
the early 1970s. Free-piston Stirling machines offer a more compact and mechanically simpler alternative to their kinematic brothers; however, these benefits come at the price of increased design complexity since the dynamics of the heat transfer directly impact the motion of the two moving elements which create the pressures – the pressures that dictate how much heat moves. In a kinematic machine, the motion of the piston/displacer is set mechanically and the thermodynamic events taking place in the machine cannot impact the displacement magnitude (or therefore, the pressure amplitude).

In either category, the free-piston or the kinematic, the energy flow in the device is the same; energy flows into the ambient temperature end (exhaust side) of the regenerator and out of the cold (load side). This amount of “left-over” energy that leaves the regenerator must be recycled to the front of the regenerator for the device to be efficient – this is another job of the mechanical linkage and flywheel of the kinematic Stirling machine, and in a free-piston machine, this energy is stored and released by the piston that resonates against the sealed volume of gas, that is, the “free” piston of the free-piston machine.

As shown in  Fig. 48.6, the innovation of Beale (or Ringbom) was to recognize that the complicated and expensive linkages found in kinematic Stirling machines could be removed and replaced with a spring; either a mechanical spring or the stiffness formed by gas in a sealed chamber. The improvement made by Backhaus, Swift, and independently by de Blok et al. was to eliminate the piston that Beale left in his device. Backhaus and Swift substituted for the piston a “slug” of gas that has relatively more inertia than the rest of the gas in the cylinder that contains the regenerator.

The terms “inertance” and “compliance” are terms that describe sections of a gas-filled resonator that behave as acoustic analogs (in a complex impedance sense) to an electrical inductor and capacitor, respectively. A necked-down section of resonator has the effect of delaying the oscillating flow of gas through it in response to a pressure difference across the section. Once the gas is accelerated by this pressure difference, it continues to move even though the pressure difference has changed direction. In order to connote this function, the word “inertance” has both the sense of “inductor” and “inertia.” A resonator section that has substantially more volume than the rest of the resonator acts like a capacitor because just as the capacitor stores energy in the form of electrical charge temporarily, the compliance section of a resonator stores energy in the form of pressure. These pressure changes lag the flow of gas into or out of the section just like the voltage across a capacitor lags the flow of current. This equivalent circuit analog to acoustical systems is quite convenient for the design of acoustic networks and is an established analytical tool in the field of acoustics [13].

The crux of Backhaus and Swift’s 1999 engine is the concept of an acoustic phasing network that both delivers acoustic power to the regenerator at a high impedance and correct phase and then returns the power that is not utilized to move heat. (In the case of an engine, the amount of power that is not taken from the wave to do work on a load is returned to the regenerator.) The function of the phasing network is twofold: condition the impedance of the incoming acoustic energy so that it can be efficiently utilized by the regenerator and then return unutilized energy back to the regenerator.



■ Fig. 48.6

The broad steps in the evolution of mechanisms to support a Stirling cycle are shown in this figure as refrigerators (these topologies are equivalent in the case of an engine). The kinematic machine, which was born in the early nineteenth century, has many forms and derivatives; only the simplest alpha configuration is shown here. The free-piston Stirling concept started to be developed in about 1970, and although the acoustic-Stirling idea was conceived of in the late 1970s, the first working prototypes were produced in the late 1990s by de Blok in the Netherlands and by Backhaus and Swift at Los Alamos National Laboratory

Future Directions (or Motivation for Development of Thermoacoustic Machines)

Now that some historical and theoretical background has provided a context for thermoacoustic machines, it is important to describe the motivation for pursuing thermoacoustic-Stirling technology.

Thermoacoustic Machine Replacements to Vapor Compression Refrigerators

A candidate replacement technology for refrigeration equipment firmly established on the market must have at least three properties: increased effectiveness (efficiency), reduced cost (simplicity), or present less of a risk for legislative phaseout. In some sense, these three reasons all collapse to the issue of cost. In order for efficiency improvements to lead to lower lifecycle costs, the increase in acquisition cost must not be greater than the fuel savings due to increased efficiency. Legislation created to limit pollution is often driven by the high cost of cleanup associated with that pollution – this is the case for greenhouse gas emissions. Reducing the carbon dioxide in the atmosphere is either very costly or not even technically well understood (an understanding that might be attainable for a higher cost than society has been willing to spend).

Several thermoacoustic machines have been built to perform low-lift (small temperature difference) cooling. One such machine, pictured in [Fig. 48.7](#), was built for Unilever's subsidiary Ben and Jerry's Homemade. This machine had a performance equivalent to the ice-cream storage cabinets that it was intended to replace but used helium as a refrigerant which has no global warming potential. The unit that it was designed to replace uses R134a (1,1,1,2-Tetrafluoroethane) which has a global warming potential (GWP) of greater than 1,400 over 100 years.



■ Fig. 48.7

A view of the regenerator-based thermoacoustic ice-cream storage cabinet built at Penn State University with sponsorship from Unilever/Ben & Jerry's. The thermoacoustic cooler is the silver unit to the *right* of the off-the-shelf storage cabinet. Heat was moved out of the storage cabinet and to the oscillating helium in the thermoacoustic machine by circulating ethanol within the copper tubing of the cabinet that typically carries R134a refrigerant and into the thermoacoustic machine's internal load heat exchanger. The photo on the *right* shows the pump that moved the ethanol covered with ice from condensed water-vapor in the air. A thermocouple reader shows the temperature of the ethanol to be -19.8°C (Left photo by Greg Grieco, used with permission)

With that in mind, there are at least three considerations that can influence lifecycle cost:

Cost to Manufacture: The commodity level vapor-compression refrigerator that is found in the kitchens of most homes in the industrialized world is inexpensive. However, since a thermoacoustic machine has few moving parts, requires no exotic materials, and has no inherently close dimensional tolerances, a commodity thermoacoustic machine stands some chance of being manufactured as inexpensively as a vapor compression machine. It is true that thermoacoustic machines, as do all closed-cycle machines, require some secondary heat exchange mechanism on both the hot and cold sides. This subsystem does increase complexity, but for some applications like fountain beverage vending, this fluid loop is integral since it is the product delivery system. In grocery store display cases and geothermal heat pumps, secondary heat transfer systems are already in use with conventional compressor technology.

Cost to Operate: The cost of energy purchased to operate the refrigeration machine is minimized by increasing its efficiency and reducing maintenance costs. Cost of energy is important for some applications where the energy is hard to transport to the location of the machine, or applications that demand significant amounts of cooling, and, therefore, reducing the energy demands of the machine saves a significant amount of money. On the other hand, the efficiency of a cooling machine is only a secondary metric for some applications; in the market of commercial ice-cream storage cabinets, the company who makes/markets the ice-cream usually provides a storage freezer to a grocery/convenience-store for no charge (as long as their product continues to sell at the store) and the shop keeper pays the operating costs. In this case, efficiency may be of secondary importance to the purchaser compared to acquisition cost. For large capacity applications such as grocery stores and refrigerated warehouses, efficiency becomes more important since the energy and acquisition cost is borne usually by the same organization and the electrical power consumption is large. Thermoacoustic machines can be made to be competitively efficient and, like Stirling machines, may become competitive with vapor compression machines. However, vapor-compression refrigeration can be made to be quite efficient if cost is no object. Efficiency of vapor-compression machinery on the market is only constrained by the acquisition price tolerance of that market. Typically, reduced operating costs, even if that reduction makes the lifecycle costs smaller, do not offset a high acquisition cost as a priority driving the purchasing decision of many consumers in both the residential and commercial sectors.

Cost to Dispose: Thermoacoustic machines use no environmentally harmful working fluids/gases. Consequently, disposal is inexpensive compared to vapor compression machines which currently utilize gases 2,000–3,000 times more potent as “green-house” gases than CO_2 . Since the machine uses no lubricants, maintenance is expected to be small and lifetime should be long. Compared to large vapor compression machines which have compressors and rotating machinery that require maintenance and suffer breakdown of the seals, thermoacoustic technology with its linear motors

and flexure or clearance seals can increase the useful lifetime of refrigeration equipment with decreases the costs of disposal and wasted material/labor.

Pitting thermoacoustic technology head-to-head with the established vapor compression technology (which is 100 years old) is daunting for as young of a technology as thermoacoustics, which has no established supplier base, a short design history and limited expertise. Vapor-compression is such a mature technology and the market has not yet found vapor compression to be deficient in the three categories above. However, if environmental legislation were to be enacted, either banning certain refrigerants or forcing consumers to pay a premium for buying and disposing of a machine that contains a global warming gas, thermoacoustic technology could become more attractive in the “cost to buy/manufacture” and the “cost to dispose” categories. As the direct cost of energy increases (direct cost is exclusive of the indirect costs such as those associated with natural resource depletion, military protection of oil prices, coal mine runoff and subsidence, etc.) due to pollution taxes or carbon credits as specified by the Kyoto Accords, thermoacoustics with its high inherent efficiency might also be attractive in the cost-to-operate category. Support for technologies that do not use global warming gases is much stronger in Europe at this time than it is in North America.

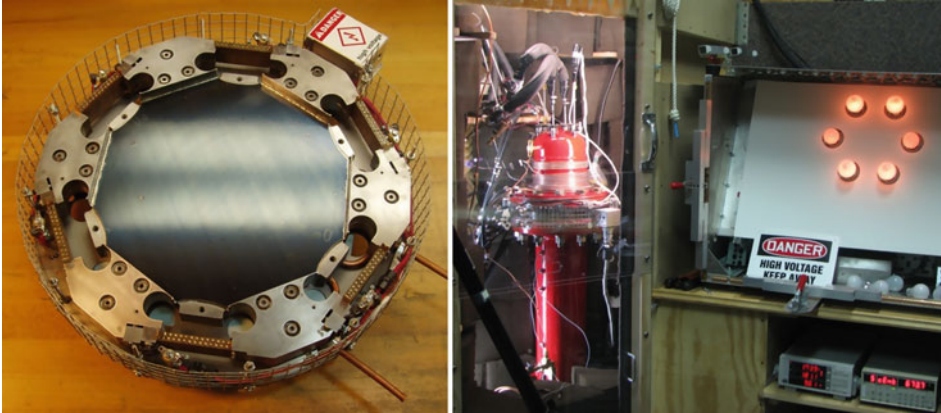
Thermoacoustic Machines as Power Generators

There are two primary applications for acoustic engines: those that convert the acoustic power to electricity or those that use the acoustic power to drive a thermoacoustic refrigerator/heat pump (there is not much demand for energy in acoustic form, and the “one-note samba” of an acoustic machine is hard to dance to). Thermoacoustic machines that are built to convert heat energy to acoustic energy and then to electrical energy might be an attractive commercialization target for flexible-fuel, small-scale power generation between 5 and 500 kW of electrical output. In-home electrical generators that burn natural gas or fuel oil to provide both the electrical power for a home and heat the home and domestic water supply might allow a household to reduce the number of subscribed utilities by at least one and eliminate the significant transmission loss associated with electrical power delivery that is wasteful of the global natural resources.

Alternators for Power Generation

A significant component in a thermoacoustic machine that converts heat to electricity is the alternator that performs the conversion of the acoustic energy to electrical energy. Carefully engineered linear motors made for this purpose are commercially available today (QDrive Corporation in Troy, New York); these are the type employed to create the sound wave that powered the thermoacoustic ice-cream freezer shown above.

However, since the diameter of the thermoacoustic machine scales with power, at high powers, the linear alternator/motor and the pistons that attach to them will have to be



■ Fig. 48.8

Two photos show the piezoelectric alternator. On the *left*, the bare alternator is shown, and the piezoelectric elements are the six tan-colored rectangular prisms arrays on the perimeter of the alternator which has an overall diameter of 35 cm. As the oscillating pressure flexes the diaphragm, the piezoelectric elements are squeezed and produce electrical power. The diaphragm squeezes the piezoelectric elements when it moves up and down. On the *right*, the alternator is installed into a regenerator-based thermoacoustic engine [15] and is powering six light bulbs with 37 W of electricity. This proof-of-concept alternator is capable of generating over 600 W of electricity (Photos by Robert Keolian, used with permission)

large and heavy, which will create a challenge for high-frequency (and therefore high power density) operation. The recent invention of a piezoelectric alternator [14] to enable power generation looks promising. Several advantages of an alternator of this design are that the piezoelectric material itself is robust, relatively inexpensive and low maintenance. Since the design of the alternator allows the piezoelectric ceramic elements to be squeezed twice per cycle by the acoustic oscillation, the power density of the alternator is much higher than that of a linear motor. ● *Figure 48.8* shows the bare alternator (left) and the alternator installed in a thermoacoustic engine generating 37 W of electricity. This photo was taken at an early stage of the development; this proof-of-concept alternator is capable of generating over 600 W of electricity.

This alternator was designed with the intention of including it in a system to capture the heat from the exhaust pipe on over-the-road trucks and convert it to electricity for use in the truck. A thermoacoustic electrical generator was designed and built at Los Alamos to power a space satellite [16]. For that project, a specialized electrodynamic alternator was designed and built.

Engines Used to Drive Heat Pumps (Refrigerators)

A very compelling application for thermoacoustic technology is to utilize a heat source directly for heat pumping. This is especially elegant because a thermoacoustic machine to

accomplish this is only comprised of heat exchangers and stack/regenerators – no moving parts or sliding/rotating seals are required. There are other technologies, like absorption refrigeration, that can have similar simplicity, but these machines typically use chemicals like ammonia and hydrogen instead of helium. They also have to work around the phase change temperatures and pressures – phase change systems have a power density advantage over non-phase change systems, but this comes at some reduced flexibility for variable working temperatures.

Along with thermoelectrics and organic Rankine cycles, thermoacoustic machines are often considered as an option for reclamation of waste heat. In many industrial processes, there is low (100–200 C) and medium temperature (200–400 C) heat that is often allowed to escape to the atmosphere unutilized. Often, the heat is difficult to divert into some machine or storage mechanism, but there is also a lack of products available on the market that can accomplish the task in a cost effective way. The lower temperature waste heat reclamation is especially challenging because the size of a device to accomplish the task is large and the Carnot efficiency is low. Nevertheless, the simple combination of a thermoacoustic engine that powers a refrigerator offers one possibility of such a solution.

There are ongoing efforts at Los Alamos National Labs in New Mexico and Aster Thermoakoustische Systemen in the Netherlands to exploit this kind of opportunity. At Los Alamos, an extensive program was executed with the goal of utilizing stranded natural gas reserves – that is, natural gas that is hard to access with piping networks to carry the gas to market. Several systems were built in escalating size and efficiency to burn natural gas at remote well-heads to power a thermoacoustic system that would liquefy the remainder of the gas stream for easy transport from the remote area [17]. The largest of these systems is shown in  Fig. 48.9. At Aster, a thermoacoustic heat-driven heat pump system was created to utilize waste heat streams to pump the remainder of the heat to a higher temperature so that it could be utilized in another industrial process.

Cookstoves for Developing World Power Generation

There has long been an interest on the part of developed nations to enable developing nations to share in the technological bounty enabled by heat engines. In many of these countries, fuel for cooking and heating is biomass (mostly wood), collected daily or weekly and oftentimes at great distances from population centers. The relative scarcity of biomass in many places is exacerbated by the collection of it, so the distance of these daily/weekly journeys increases at alarming rates through difficult and/or dangerous terrain. Additionally, the relatively poor utilization of these fuels in stoves or open fires with incomplete combustion causes very poor indoor and outdoor air quality, and the health of women and children (who spend more time around the fires and in the house than men) suffers dramatically. Poor indoor air quality caused by inefficient cookstoves is as big of a health risk than any other in some developing countries, causing 1.9 million deaths each year and chronic illnesses such as low birth weight, acute pneumonia in children under 5, lung cancer, chronic obstructive pulmonary disease, and cardiovascular



■ Fig. 48.9

A photo of the natural gas liquifier next to a six-foot (1.8 m) tall man shows the highest power thermoacoustic device ever built. At the top is a blower attached to the combustion chamber. The thermoacoustic engine is below the combustion chamber (the conical shaped component) and is connected *via* the wave tube to a vacuum can at the bottom next to John Wollan, one of the designers and builders of the machine. This device liquified 350 gal/day with a projected production efficiency of 70% liquifaction and 30% combustion of an incoming gas stream (Photo by Greg Swift, used with permission)

disease [18]. From the standpoint of global warming, the billions of inefficient, poorly constructed cookstoves in use today contribute to global warming through not only the CO_2 , methane, and nitrous oxide present in the combustion products, but also the soot or black carbon emissions from these stoves. Particulate matter of elemental carbon is suspect as a major contributor to climate change and has an estimated GWP of 680 [19].

It has been established [20] that a fan integrated into the combustion chamber of a stove can increase combustion efficiency and dramatically decrease particulate and black carbon emissions. Lowered emissions can improve atmospheric health in a very short timeframe, and increased efficiency translates to less deforestation and increased carbon sequestration in these areas. Since one way to power a fan is with electricity, there has been

some interest to utilize a thermoacoustic electrical generator to provide this electricity using a small fraction of the heat produced by the stove's fuel. Cost is one of the dominant constraints for the successful adoption of technology in the developing world, and the mechanical simplicity of thermoacoustic machines, the lack of exotic materials, and no need for tight machining tolerances all make thermoacoustic technology a candidate for small-scale power generation in developing countries. The other likely technology to fill this niche, which has undergone more testing and commercialization for the application than thermoacoustics, is thermoelectric modules.

Aster Thermoakoestische Systemen has recently completed the testing of a prototype intended for this market with funding from the SCORE program at the University of Nottingham (SCORE stands for Stove for Cooking, Refrigeration and Electricity) and the FACT Foundation in Eindhoven, Netherlands. The topology of the thermoacoustic engine designed and built by Aster is novel [21] due to its planar geometry shown in [Fig. 48.10](#) which allows integration to a stove by presenting a large heat exchange area to the hot gases above the fire. The thermoacoustic machine is a four stage regenerator-based engine coupled to electro-dynamic alternators. Each of four regenerators is positioned on a set of ambient-side fins shown below on the left hand side of [Fig. 48.10](#). The tubes connecting each of the four quadrants are part of the acoustic network. A similar finned plate is installed on the ambient plate which sandwiches the regenerators between the plates and making a series connection between each of the four stages of the machine. This ambient fin plate is shown immersed in the water bath on the right-side photo of [Fig. 48.10](#) and the hot-side fin plate is facing the gas burner. In practice, the burner could be a three-stone fire or cookstove. The electro-dynamic alternators for this design are still under-construction.



■ Fig. 48.10

The ambient temperature heat exchange plate of the cookstove generator developed by Aster is shown on the left-hand side and the full machine is shown on the right being fired by a natural gas burner. A pot of water serves as the exhaust-side sink; the device is intended to provide hot water and electricity for citizens of developing countries (Photo by Kees de Blok, used with permission)

Commercialization Considerations

Direct competition with vapor compression technology may not be the only path toward commercialization of thermoacoustic cooling technology. There is surely an application for a machine that uses an environmentally benign working fluid, is quiet, efficient, has low moving mass, and can easily take advantage of proportional control that could propel the technology to prominence even at a cost premium. Consequently, putting research efforts into increasing the power density, reducing cost, and increasing efficiency is not wasted – especially in light of the fact that thermoacoustic technology is functionally reversible. This means that knowledge gained by research into the machine as a refrigerator is directly transferable to the design of a machine that is built as an engine.

Now that a clear link has been established between the operation of thermoacoustic technology and Stirling machines, the next logical question is “Why spend money/time developing thermoacoustic machines in favor of developing Stirling machines?” Alternatively, this question could be asked of a developer of thermoacoustic technology: “What makes you think that thermoacoustic machines will be commercially feasible when Stirling machines have not reached a ‘commodity status’ during the nearly 200 year history of their development?” (Kinematic Stirling machines attained a brief “commodity status” at the start of the twentieth century as water pumps and compressors.) There are several clear advantages and some disadvantages of thermoacoustics when compared to Stirling (► [Fig. 48.11](#)).

Mechanical Complexity: Stirling machines, and kinematic Stirling machines especially, are mechanically complex. A kinematic machine requires some mechanical means to link the pistons in such a way to create the correct phasing of pressure oscillation and gas displacement. In ► [Fig. 48.6](#), two kinematic machines are shown that were made at the extremes of the twentieth century, and both are burdened with this linkage that requires lubrication, experiences wear, dissipates energy, and is costly to manufacture and maintain. Many Stirling machines, both kinematic and free-piston, have a piston (to create pressure oscillations) and a displacer (to move the gas through the regenerator at a roughly constant volume) that must move in the same bore. In many designs, this requires that the rod for the displacer penetrate the piston and sometimes that both of the connectors for these mechanical components penetrate the pressure vessel that contains the working fluid (usually helium). These dynamic seals, sometimes in the presence of either hot or cold temperatures, present a difficult engineering problem that is usually costly to implement due to close-tolerance manufacturing requirements that provide simultaneously low friction and low leakage. The most significant contribution of thermoacoustic technology to the development of regenerative heat engines is simplicity. By eliminating the displacer or second piston (and in a prime mover, by removing all moving metal) in favor of gas inertia and by exchanging a mechanical phasing mechanism for one that is executed using the compliance and inertia of the working fluid itself, a much simpler and consequently cheaper machine can be made. Thus far, this reduction of complexity has probably come at the price of power density. In the machines that are the subject of this chapter, the sliding seals that dominate Stirling machines have been



■ Fig. 48.11

On the *left* is photograph of a Rider-Ericson engine produced in the late 1800s and early 1900s. Notice the complicated linkage that connects the flywheel to the piston rods that penetrate the *top* of the piston cylinder. On the *right* is a modern Stirling machine made by Whisper Tech (<http://www.whispergen.com/>) in 2002 that is used to generate electricity for residences, yachts, and motor-homes. Hidden in the stainless steel pressure vessel is a complicated linkage called a “wobble-yoke” (*left* photo in the public domain, *right* photo credit: Whispergen)

replaced by flexure seals. These seals, while simple and low maintenance require careful design for “infinite” fatigue life. At this time, the metal bellows that are used are expensive.

Power Density: The power density of thermoacoustic machines tends to be neither as high as that of its Stirling “brothers” nor as high as either vapor compression refrigerators or internal combustion engines. The relatively low power density comes as consequence of relying on the stiffness (compliance) of gas trapped in the resonator to provide restoring force and moving gas to provide inertia. Compared to a mechanical spring, gas springs are much bigger for a given stiffness and the density of gas is much lower than the density of metal or plastic used as pistons in traditional Stirling machines. That said, since the compliance of trapped gas is a function of the mean pressure in the gas ($C_g = V/\gamma p_m$) and since density is also proportional to mean pressure, by increasing the mean pressure in a thermoacoustic machine, the power density can be improved, although the cost/weight/complexity of a stronger pressure vessel must be considered.

Moving Mass: The moving mass of thermoacoustic machines can be low for a given amount of cooling capacity (or power output if operating as an engine) compared to

Stirling machines. This has advantages with respect to fatigue life as well as transmitted noise/vibration levels and radiated noise levels.

Resonance: One overarching advantage of thermoacoustic machines described herein is that since the energy storage medium is the working gas itself and the moving mass, there are no requirements for external energy storage elements (like flywheels) that are found in many traditional Stirling machines.

Limitations Imposed by Heat Exchangers: All closed-cycle engines suffer an efficiency penalty compared to open-cycle machines like internal combustion engines or vapor-compression refrigerators. This penalty comes from the extra heat-exchange step inherent in closed-cycle engines because the working gas does not directly contact the heat source and sink. In the internal-combustion engine, for example, the fuel enters the engine cylinder and explodes – the engine takes advantage of the very hot temperature of that explosion in the form of the expansion of the combustion products. Following that expansion, the hot gas is exhausted from the cylinder, whereupon a new charge of cool fuel/air mixture is injected. Because of this rapid exchange of hot gas with cool gas and the fact that the hottest temperature of the explosion only lasts for an instant, the metal of engine block does not have to sustain the high temperatures of which the engine gets to take advantage. Closed-cycle engines usually have to be designed for significantly lower temperatures due to material constraints, but they also suffer a temperature difference across the heat exchanger through which all of the heat that the engine uses must pass. As a result, the engine takes an efficiency penalty because it cannot utilize the highest temperature that the fuel is providing.

Temperature-related material constraints are not usually a factor in a heat pump application, but the same handicap arises due to the temperature drop on the heat exchangers that transfer heat from the cooling load and again out of the machine into the environment. Especially for low-lift applications (small temperature difference) like food refrigeration or air-conditioning, the temperature drop on the primary helium-to-secondary heat exchangers can be a significant fraction of the total temperature difference and therefore be the largest efficiency penalty in the system. In an open-cycle vapor compression system, where the refrigerant undergoes a phase change, the compressor effectively pumps the refrigerant between two heat exchangers that directly exchange heat with the load and exhaust. Consequently, these systems have half of the heat exchange steps than a closed-cycle machine (like one that utilizes thermoacoustics) would have. The designer of a thermoacoustic machine that has to compete only in terms of efficiency has to carefully consider the primary (oscillating gas to secondary fluid) and secondary (secondary fluid to load/environment) heat exchangers. Ideas to eliminate the secondary heat exchange step have been explored with some success; for example, a design was tested at Los Alamos that acoustically pumped helium out of the resonator to the heat source/sink [22]. A flow-through design was also tested at Los Alamos that allowed a steady flow of air to move through the resonator; this machine used air at atmospheric pressure as the working gas and, therefore, was not particularly power dense in either weight or volume [23].

References

1. Ceperley PH (1979) A pistonless Stirling engine – the traveling wave heat engine. *J Acoust Soc Am* 66(5):1508–1513
2. Wheatley JC, Swift GW, Migliori A (1983) Acoustical heat pumping engine. US Patent 4,398,398, 16 Aug 1983
3. Swift GW (2002) Thermoacoustics: a unifying perspective for some engines and refrigerators. Acoustical Society of America, New York
4. Swift GW (1988) Thermoacoustic engines. *J Acoust Soc Am* 84(4):1145–1180
5. Liu J, Garrett SL (2005) Characterization of a small moving-magnet electrodynamic linear motor. *J Acoust Soc Am* 118(4):2289–2294
6. Kolin I (1995) Thermodynamic theory for Stirling cycle machine designs: special lecture 1. In: Proceedings of the seventh international conference on Stirling cycle machines, Waseda University, Tokyo, pp 1–7
7. Urieli I, Berchowitz D (1984) Stirling cycle engine analysis. A. Hilger, Bristol
8. Sier R (1995) Rev. Robert Stirling D. D.: inventor of the heat economiser and Stirling engine. Ipswich Book, Ipswich
9. Wheatley JC, Swift GW, Migliori A (1986) The natural heat engine. *Los Alamos Sci* 14:2–33, LAUR 86–2699
10. Backhaus S, Swift GW (2000) A thermoacoustic-Stirling heat engine: detailed study. *J Acoust Soc Am* 107:3148–3166
11. de Blok, Maria C, Hendrikus NA, Van Rijt J (2001) Thermo-acoustic system. US Patent 6,314,740
12. Beale WT (1971) Stirling cycle type thermal device. US Patent 3,552,120
13. Beranek LL (1954) Acoustic elements, Chapter 5. In: Acoustics. Acoustical Society of America Through the American Institute of Physics, Massachusetts, 116–143
14. Keolian RM, Wuthrich JW, Bastyr KJ (2007) Thermoacoustic piezoelectric generator. US Patent 77, 72, 746
15. Bastyr KJ (2004) The design, construction, and performance of a high-frequency, high-power thermoacoustic-Stirling engine. Doctoral dissertation, The Pennsylvania State University
16. Backhaus S, Tward E, Petach M (2004) Traveling-wave thermoacoustic electric generator. *Appl Phys Lett* 85:1085–1087
17. Bayram A et al (2005) Operation of thermoacoustic Stirling heat engine driven large multiple pulse tube refrigerators. In: Ross RG Jr (ed) Cryocoolers 13: proceedings of the 13th ICC held in New Orleans, Louisiana. Springer Science + Business Media, New York
18. Global Alliance for Clean Cookstoves (2010) National Institute of Environmental Health Science, National Institutes of Health. <http://www.niehs.nih.gov/about/od/programs/cookstoves/index.cfm>. Accessed 17 Mar 2011
19. Roden C, Bond T (2006) Emission factors and real-time optical properties of particles emitted from traditional wood burning cookstoves. *Environ Sci Technol* 40(21):6750–6757
20. MacCarty N et al. (2008) A laboratory comparison of the global warming impact of five major types of biomass cooking stoves. *Energy Sust Dev* 12:56–65
21. de Blok K (2010) Novel 4-stage traveling wave thermoacoustic power generator. In: Proceedings of ASME 2010 3rd Joint US-European fluids engineering summer meeting and 8th international conference on nanochannels, microchannels and minichannels, Montreal
22. Swift GW, Backhaus S (2004) A resonant, self-pumped, circulating thermoacoustic heat exchanger. *J Acoust Soc Am* 116(5):2923–2938
23. Reid RS, Swift GW (2000) Experiments with a flow-through thermoacoustic refrigerator. *J Acoust Soc Am* 108(6):2835–2842